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EFFECT OF REDUCING ROTOR BLADE INLET DIAMETER ON THE PERFORMANCE OF A 11.66-CENTIMETER RADIAL-INFLOW TURBINE

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16. Abstract

The effect of increased rotor blade loading on turbine performance was investigated by reducing rotor blade inlet diameter. The reduction was made in four stages. Each modification was tested with the same stator using cold air as the working fluid. Results are presented in terms of equivalent mass flow and efficiency at equivalent design rotative speed and over a range of pressure ratios. Internal flow characteristics are shown in terms of stator exit static pressure and the radial variation of local loss and rotor-exit flow angle with radius ratio. Included are velocity diagrams calculated from the experimental results.

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EFFECT OF REDUCING ROTOR BLADE INLET DIAMETER ON THE PERFORMANCE OF A 11.66-CENTIMETER RADIAL-INFLOW TURBINE

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SUMMARY

An experimental investigation was made to determine the effect of increased blade loading on performance by reducing the rotor blade inlet diameter on a 11.66-centimeter (4.59-in.) radial-inflow turbine. Four reductions were made in blade inlet diameter to 11.05, 10.41, 9.40, and 8.38 centimeters (4.35, 4.10, 3.70, and 3.30 in.). Turbine performance was measured using the same stator with each modified rotor. Tests were made at equivalent design rotative speed and over a range of pressure ratio from 1.28 to 1.93. All tests were conducted with cold air as the working fluid.

Results of the investigation indicated that maximum efficiencies were obtained with a clearance ratio of 0.075 for this turbine. Clearance ratio is defined as the difference between the diameter at the stator blade trailing edge and the rotor blade inlet diameter divided by the diameter at the stator blade trailing edge. The total and static efficiencies for this clearance ratio were 0.90 and 0.84 at design equivalent rotative speed and pressure ratio. Reduced rotor blade wetted area and reduced rotor losses resulted in the improvement in performance over that obtained with the 11.66-centimeter or design rotor configuration. The maximum peak total and corresponding static efficiencies were 0.91 and 0.85. These values were obtained at design equivalent rotative speed and at a pressure ratio lower than the design value of 1.54. This improvement in peak efficiencies over those obtained at design pressure ratio was attributed to the lower rotor incidence losses. The poorest performance was obtained with the 8.38-centimeter rotor configuration. The poor performance resulted primarily from increased rotor losses due to high blade loading with possible flow separation.

Equivalent mass flow decreased with decreasing rotor blade inlet diameter for turbine inlet total to rotor exit static pressure ratios of design value and larger. At design pressure ratio, there was a 6.1-percent decrease in mass flow for a 28.1-percent decrease in rotor blade inlet diameter. This decrease in mass flow at design pressure ratio was the result of an increase in rotor blade incidence and blade loading losses.

INTRODUCTION

Small radial-inflow turbines have been experimentally investigated at the Lewis Research Center in order to determine the factors that influence their performance. This work included the investigation of the effect of volume flow (specific speed) on turbine performance. Variation in volume flow was made by changing the stator and rotor blade throat areas. Results are reported in references 1 and 2. The effect of turbine size on performance is reported in reference 3. In this reference experimental results of two smaller scale versions of a 15.29-centimeter (6.02-in.) diameter turbine were compared with those of the full-scale turbine. The results of varying the blade-shroud clearance on the 15.29-centimeter (6.02-in.) diameter turbine are reported in reference 4. The isolated effects of axial and exducer radial clearance on efficiency were determined. It was found that the exducer radial clearance was more influential by a factor of 10 in the clearance range from near 0 to 8 percent of blade height. The 11.66-centimeter (4.59-in.) turbine of reference 1 was used to investigate the effect of increased rotor blade loading by removing the rotor splitter vanes. The results, as reported in reference 5, showed no decrease in performance when the splitter vanes were removed. These tests, together with the tip clearance investigation of reference 4, indicated that radial turbines are relatively insensitive to unfavorable flow conditions in the upstream part of the rotor flow passages.

In view of this, it was of interest to determine the effect of further increases in blade loading on performance by reducing rotor blade inlet diameter. The reduction of rotor blade inlet diameter has distinct advantages. Stator blade erosion can be reduced by the increased radial clearance between the stator blade trailing edge and the rotor inlet. Rotor blade tip speed is reduced thereby reducing the stress level. In addition, the increased radial distance between the stator trailing edge and the rotor blade inlet will result in more complete mixing of stator blade wakes with the mainstream flow. The turbine of reference 5 was used for the extension of the program on increased rotor blade loading. Reduction in rotor blade inlet diameter was made in four stages, resulting in diameters of 11.05, 10.41, 9.40, and 8.38 centimeters (4.35, 4.10, 3.70, and 3.30 in.). These values represent a 5.2, 10.7, 19.4, and 28.1 percent reduction in diameter from that of the original rotor. Each turbine configuration consisting of the design stator and the modified rotor was experimentally investigated at equivalent design rotative speed and over a range of pressure ratios. Air was used as the working fluid.

This report presents the effects on efficiency and mass flow of increasing the blade loading by reducing the rotor blade inlet diameter (increased blade loading) for a given initial radial turbine configuration. Results are presented in terms of equivalent mass flow and efficiency at equivalent design rotative speed and over a range of pressure ratios. Internal flow characteristics at equivalent design rotative speed and design pressure ratio as well as at the pressure ratio corresponding to maximum efficiency are

used to explain the changes in performance. The internal flow characteristics are presented in terms of stator exit static pressure, radial variation of local loss and rotor exit flow angle, and velocity diagrams.

SYMBOLS

A area, cm^2; $in.^2$

D_R diameter at rotor blade leading edge, m; ft

D_S diameter at stator blade trailing edge, m; ft

g dimensional constant, SI = 1.0; 32.174 ft/sec^2

Δh specific work, J/g; Btu/lb

N rotative speed, rpm

p absolute pressure, N/cm^2; psia

R gas constant, J/(kg)(K); ft-lb/(lb)($^{\circ}R$)

r radius, m; ft

T absolute temperature, K; $^{\circ}R$

U blade velocity, m/sec; ft/sec

V absolute gas velocity, m/sec; ft/sec

V_j ideal jet-speed corresponding to total- to static-pressure ratio across turbine, m/sec; ft/sec

W relative gas velocity, m/sec; ft/sec

w mass flow, kg/sec; lb/sec

α absolute rotor exit gas flow angle measured from axial direction, deg

γ ratio of specific heats

δ ratio of inlet total pressure to U.S. standard sea-level pressure, p_1'/p^*

ϵ function of γ used in relating parameters to those using air inlet conditions at U.S. standard sea-level conditions, $(0.740/\gamma)\left[(\gamma + 1)/2\right]^{\gamma/(\gamma-1)}$

η efficiency

η_t total efficiency (based on inlet-total to exit-total pressure ratio)

θ_{cr} squared ratio of critical velocity at turbine-inlet temperature to critical velocity at U.S. standard sea-level temperature, $\left(V_{cr}/V_{cr}^*\right)^2$

In the calculation of turbine inlet total pressure, flow angle was assumed to be zero.

Data were obtained at nominal inlet total conditions of 306 K (551° R) and 10.34 newtons per square centimeter (15.00 psia). Tests were made at equivalent design rotative speed and over a range of inlet total to exit static pressure ratios of 1.28 to 1.93.

Two radial surveys were made at the rotor exit to determine the variation of exit flow angle, total pressure, and total temperature with radius. One survey was made at equivalent design rotative speed and at design total to static pressure ratio of 1.54. The other survey was made at equivalent design rotative speed and at the total to static pressure ratio corresponding to maximum total efficiency

The velocity diagrams, shown in this report, were determined from speed, work, mass flow, flow angles, pressure measurements at turbine inlet and rotor exit, turbine inlet total temperature, and stator and rotor throat areas.

RESULTS AND DISCUSSION

An experimental investigation was made to determine the effect of increased rotor blade loading on turbine performance by reducing rotor blade inlet diameter. Rotor blade inlet diameter was reduced by removing rotor blade material and leaving the hub disk at its original diameter. Results are presented in terms of mass flow, efficiency, rotor exit flow angle, stator exit static pressure, and turbine loss. In addition, turbine velocity diagrams are shown for each configuration at the 50 percent streamline, at equivalent design rotative speed and pressure ratio, and at the pressure ratio corresponding to maximum total efficiency. The design rotor and subsequent rotors of reduced blade inlet diameters will be referred to as the 11.66-, 11.05-, 10.41-, 9.40-, and 8.38-centimeter configurations.

Turbine Performance

Figure 3 shows the variation of mass flow with turbine inlet total to rotor exit static pressure ratio. The figure indicates that mass flow decreased substantially with decreasing rotor blade inlet diameter for pressure ratios larger than the design value. This decrease in mass flow, at a given pressure ratio, was the result of increased rotor losses. For a given pressure ratio, rotor incidence losses would be expected to increase substantially with decreasing rotor blade inlet diameter. The increased rotor losses, at a given pressure ratio, would be reflected in an increase in stator exit static pressure and hence a decrease in mass flow. There was a 6.1-percent decrease in mass flow for a 28.1-percent decrease in rotor blade inlet diameter at design pressure ratio. It will be noted that the same mass flow was obtained for the 11.66- and 11.05-

centimeter configurations at design pressure ratio. This would indicate that the rotor losses were about the same for both configurations. The effect of rotor inlet diameter on mass flow decreased significantly as the pressure ratio decreased.

Figure 4 presents turbine efficiency as a function of blade-jet speed ratio for the five rotor configurations investigated. Total to static pressure ratios (p_1'/p_3) corresponding to their respective blade-jet speed ratios are shown by the vertical dashed lines. The design pressure ratio was 1.54. The maximum peak total and corresponding static efficiencies of 0.91 and 0.85, respectively, were obtained at a pressure ratio of 1.40 for the 11.05-centimeter configuration (fig. 4(b)). The improvement in efficiency over the design configuration (fig. 4(a)) (total and static efficiencies of 0.89 and 0.84, respectively), can be attributed to reduced rotor blade surface area and reduced overall rotor losses. Apparently these decreased losses more than accounted for any increase in loss associated with increased rotor blade loading. It will be noted that the total efficiency peaks at lower pressure ratios as the blade inlet diameter was decreased. Decreasing pressure ratio resulted in rotor incidence angles closer to optimum value as the inlet blade speed or blade inlet diameter was decreased. The lowest peak total efficiency of 0.84 and corresponding static efficiency of 0.79 were obtained with the 8.38-centimeter configuration at a pressure ratio of 1.28 (fig. 4(e)). The large decrease, from the values obtained with the 11.05-centimeter configuration, is mainly attributed to the substantial increase in blade loading, which probably results in flow separation.

Figure 4 also shows that the exit kinetic energy level, as denoted by the difference between total and static efficiencies, decreased with decreasing rotor blade inlet diameter. The decrease in kinetic energy is especially evident at the lower blade-jet speed ratios (higher pressure ratios) and results from the reduction in mass flow as shown in figure 3.

Figure 5 presents total and static efficiency as a function of clearance ratio for operation at design pressure ratio and at the pressure ratio corresponding to the peak efficiency for each rotor configuration investigated. Clearance ratio is defined as the difference between the diameter at the stator blade trailing edge and the rotor blade inlet diameter divided by the diameter at the stator blade trailing edge. The dashed vertical lines along the abscissa indicate the value of the clearance ratio for each rotor configuration investigated. The figure shows that for this turbine the maximum of the peak total efficiency curve is 0.91 at a clearance ratio of about 0.075. The maximum static efficiency from the peak static efficiency curve is 0.85. This value could also be obtained for this turbine at this clearance ratio of 0.075. The decrease in peak total efficiency with increased clearance ratio was attributed to the increase in blade loading with possible flow separation. Total and static efficiencies for the clearance ratio of 0.075 were 0.90 and 0.84 at design equivalent rotative speed and pressure ratio. The difference in total efficiency levels between the two curves (peak efficiency and efficiency

at design pressure ratio) was due mainly to the increased rotor incidence losses at design pressure ratio.

Internal Flow Characteristics

Figure 6 presents the variation of rotor exit flow angle and local loss with radius ratio at design equivalent rotative speed and pressure ratio for the five configurations investigated. Increased turning (from axial) occurred with decreasing rotor blade inlet diameter from the hub to the radius ratio of 0. 500 when compared with the 11. 66-centimeter configuration. Considerably decreased turning was obtained for the 8. 38-centimeter configuration along the tip portion of the blade. This may have resulted from some flow separation due to the high rotor blade loading.

The local loss, defined as $1 - \eta_t$, gradually increased from the inner wall to the outer wall for all rotor configurations except the design or 11. 66-centimeter configuration. Results of the survey for this configuration showed almost a constant value of loss from the hub to the radius ratio of 0. 60 and then increased sharply to the outer wall. The loss for the 8. 38-centimeter configuration was substantially higher than for the others from a radius ratio of 0. 50 to the outer wall. It would appear that the trend of exit flow angle with rotor configuration is not consistent with the trend of local loss particularly in the region of the hub to a radius ratio of 0. 7. Since the exit flow angle became larger in negative values (greater turning) with decreasing blade inlet diameter, one would expect the local loss to decrease with decreasing blade inlet diameter. Results of the radial surveys at the rotor exit indicated that the rotor inlet UV_u decreased at a greater rate, with decreasing blade inlet diameter, than the rate of increase in rotor exit UV_u. The net result was the increase in local loss with decreasing blade inlet diameter.

The variation of turbine exit flow angle and local loss at equivalent design rotative speed and at the pressure ratio corresponding to maximum total efficiency for each rotor configuration is shown in figure 7. Largest turning occurred for the 11. 66-centimeter configuration. The turning decreased with decreasing rotor inlet tip diameter. The curves of local loss as a function of radius ratio shows that the local loss increased as the blade inlet diameter was decreased. The reason for the high loss near the rotor tip for the 11. 66-centimeter configuration is not known at this time. Mass averaged local efficiencies for the 11. 66- and 11. 05-rotor configurations agreed closely with the total efficiencies obtained by torque measurements.

Figure 8 shows the variation of stator exit static pressure with rotor inlet diameter at design rotative speed and pressure ratio. The figure indicates that the stator exit velocity decreased with decreasing rotor blade inlet diameter. This was to be expected

since rotor losses increased with decreasing rotor inlet diameter and the increased loss would result in an increased stator exit static pressure.

Figure 9 shows the turbine velocity diagrams for each configuration for design equivalent rotative speed at design pressure ratio, and at the pressure ratio corresponding to maximum efficiency. The velocity diagrams were calculated from the experimental results. Both cases showed a large change in the rotor blade relative inlet angle as the blade inlet diameter was reduced. At design equivalent speed and pressure ratio the relative inlet angle changed from -26.5° for the 11.66-centimeter configuration to 47.9° for the 8.38-centimeter configuration. Design value was -23.6° for the 11.66-centimeter configuration. At the condition of maximum total efficiency the rotor relative inlet angle changed from -26.5° to 13.9°. This large change in relative inlet angle would result in excessive incidence losses and high velocity peaks on the suction surface with large values of diffusion for the 9.40- and 8.38-rotor configurations. The diagrams also indicate that there was a substantial decrease in rotor reaction $\left[1 - \left(W_2^2 \;\; W_3^2\right)\right]$ for the 9.40- and 8.38-rotor configurations when compared with the reaction obtained for the 10.41-rotor configuration at design pressure ratio.

SUMMARY OF RESULTS

An experimental investigation of a 11.66-centimeter radial-inflow turbine was conducted to determine the effect of increasing rotor blade loading by reducing rotor inlet diameter. Rotor configurations having blade inlet diameters of 11.66, 11.05, 10.41, 9.40, and 8.38 centimeters (4.59, 4.35, 4.10, 3.70, and 3.30 in.) were investigated. The results of this investigation are summarized as follows:

1. Optimum performance could be obtained with a clearance ratio of 0.075 for this turbine. Clearance ratio is defined as the difference between the diameter at the stator blade trailing edge and the rotor blade inlet diameter divided by the diameter at the stator blade trailing edge. The total and static efficiencies for this clearance ratio of 0.075 were 0.90 and 0.84 at design equivalent speed and pressure ratio.

2. The maximum peak efficiencies were also obtained for this turbine at this same clearance ratio of 0.075. The total and static efficiencies of 0.91 and 0.85 were obtained at design equivalent speed and at a pressure ratio lower than the design value of 1.54. This improvement in peak efficiencies over those obtained at design pressure ratio was attributed to smaller rotor incidence losses.

3. Operation with the 8.38-centimeter rotor configuration resulted in the poorest performance. This was attributed mainly to increased losses due to blade loading with possible flow separation.

4. Equivalent mass flow decreased with decreasing rotor blade inlet diameter for turbine inlet total to rotor exit static pressure ratios of design value and larger. There

was a 6.1-percent decrease in mass flow for a 28.1-percent decrease in rotor blade inlet diameter at design pressure ratio. This decrease in mass flow was a result of an increase in rotor incidence and blade loading losses.

Lewis Research Center,
National Aeronautics and Space Administration,
and
U.S. Army Air Mobility R&D Laboratory,
Cleveland, Ohio, November 10, 1972,
503-35.

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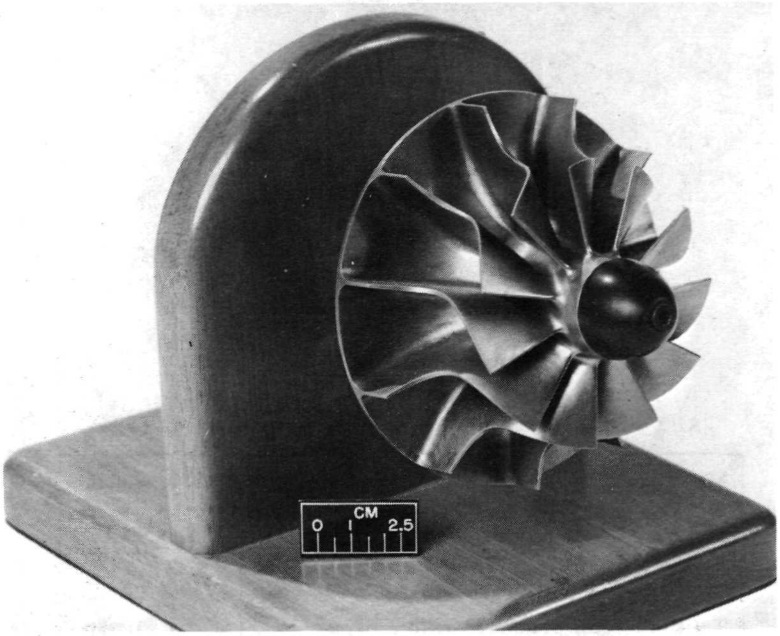
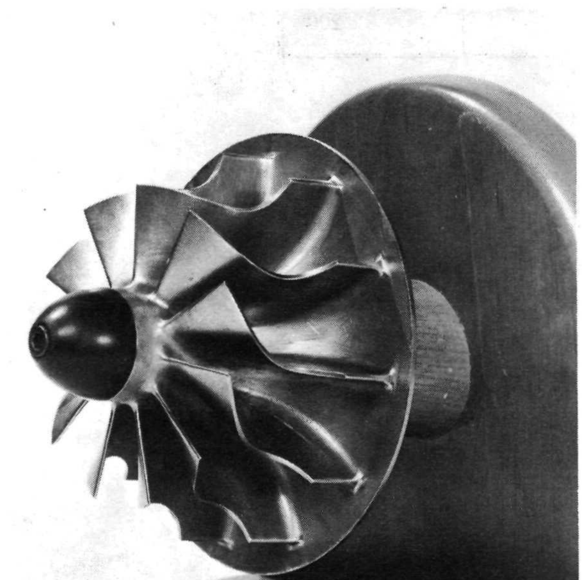
TABLE I. - AIR EQUIVALENT DESIGN PARAMETERS

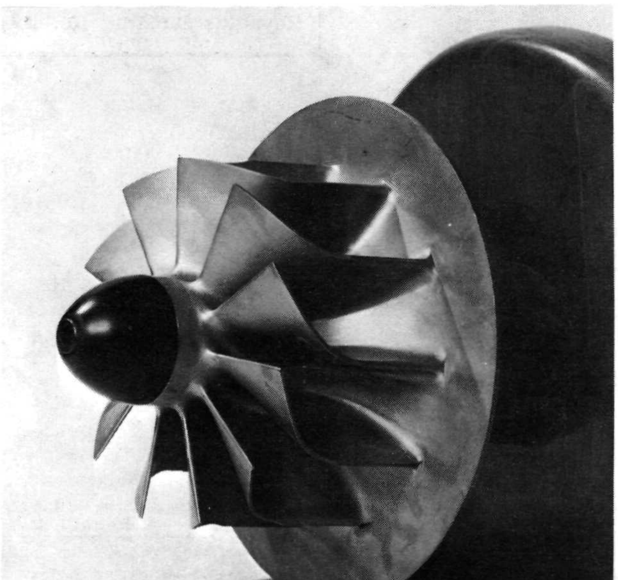
Mass flow rate, $\epsilon w \theta_{cr}/\delta$, kg/sec; lb/sec	0.279; 0.616
Specific work, $\Delta h/\theta_{cr}$, J/g; Btu/lb	27.7; 11.9
Rotative speed, $N/\sqrt{\theta_{cr}}$, rpm	29 550
Torque, $\epsilon \tau/\delta$, N-m (in. -lb)	2.50; 22.12
Total-to-total pressure ratio, p_1'/p_3'	1.496
Total-to-static pressure ratio, p_1'/p_3	1.540
Blade-jet speed ratio, ν	0.697
Reynolds number, Re, $w/\mu r_t$	82 200



(a) Blade inlet diameter, 11.66 centimeter (4.59 in.).



(b) Blade inlet diameter, 10.41 centimeter (4.10 in.).



(c) Blade inlet diameter, 8.38 centimeter (3.30 in.).

Figure 1. - Rotor.

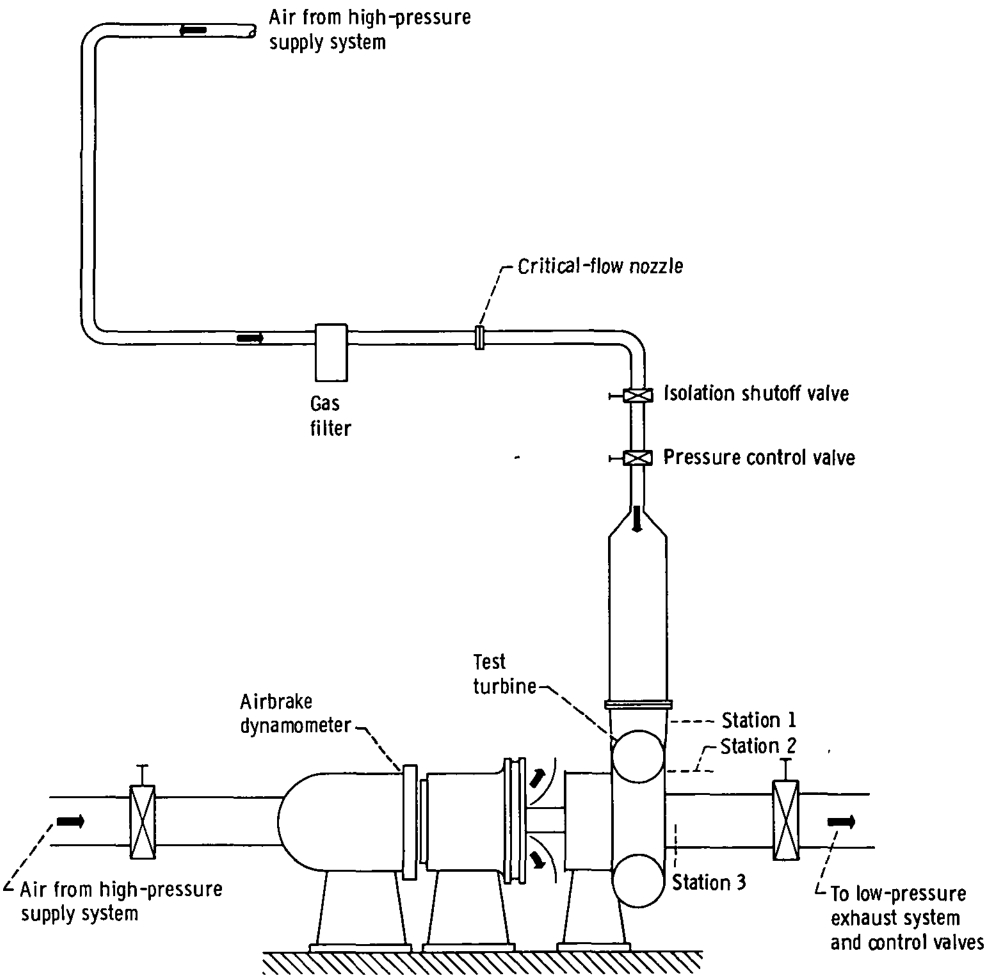


Figure 2. - Experimental equipment.

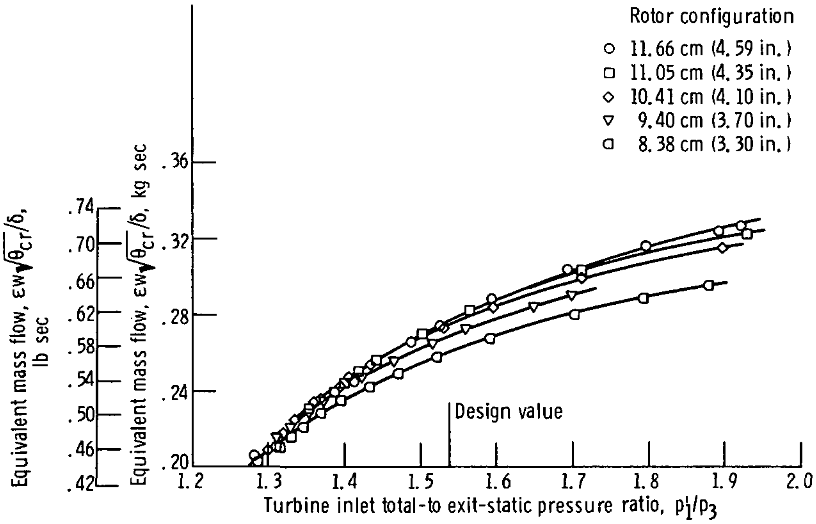
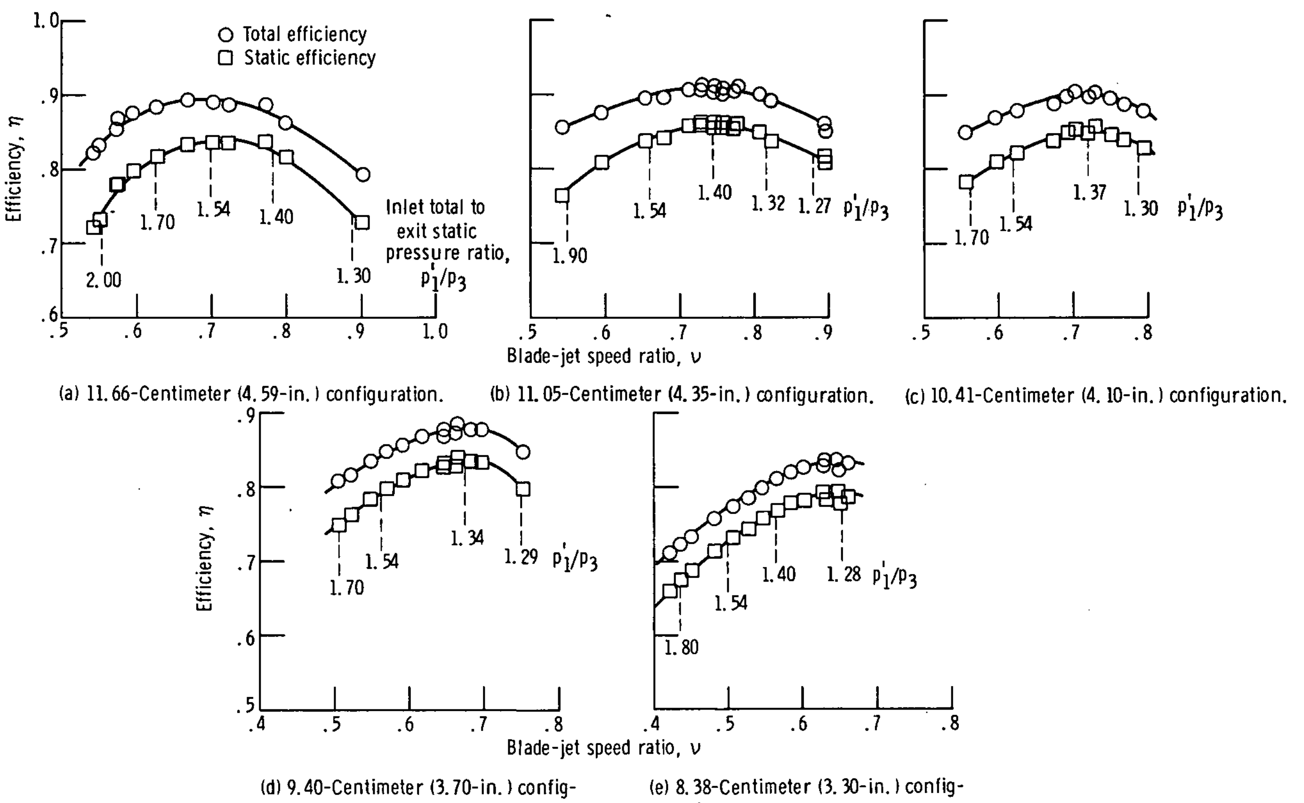


Figure 3. - Variation of equivalent mass flow with pressure ratio at equivalent design rotative speed.



(a) 11.66-Centimeter (4.59-in.) configuration.

(b) 11.05-Centimeter (4.35-in.) configuration.

(c) 10.41-Centimeter (4.10-in.) configuration.

(d) 9.40-Centimeter (3.70-in.) configuration.

(e) 8.38-Centimeter (3.30-in.) configuration.

Figure 4. - Variation of efficiency with blade-jet speed ratio at equivalent design rotative speed.

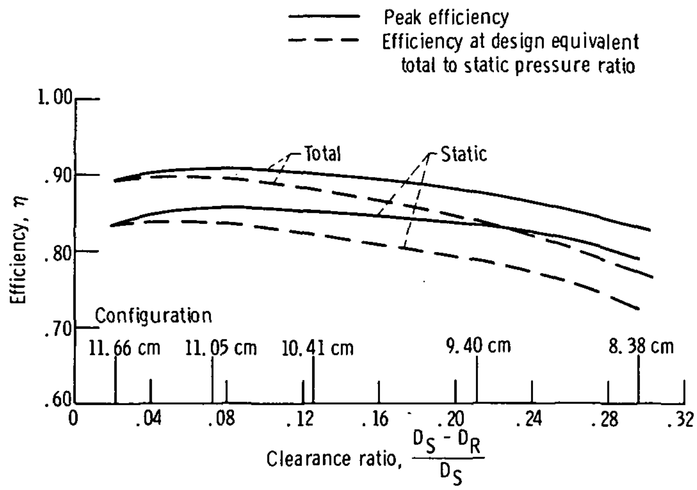


Figure 5. - Variation of efficiency with clearance ratio at equivalent design speed.

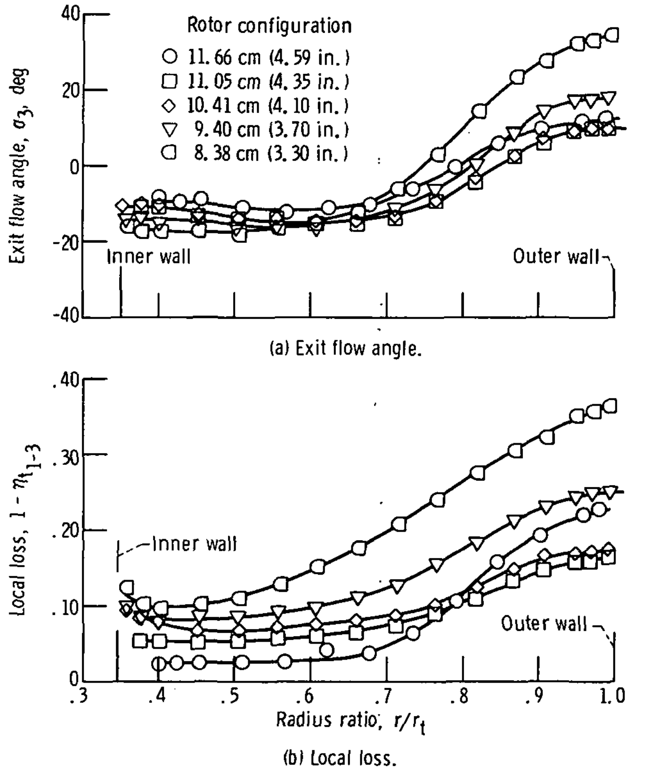
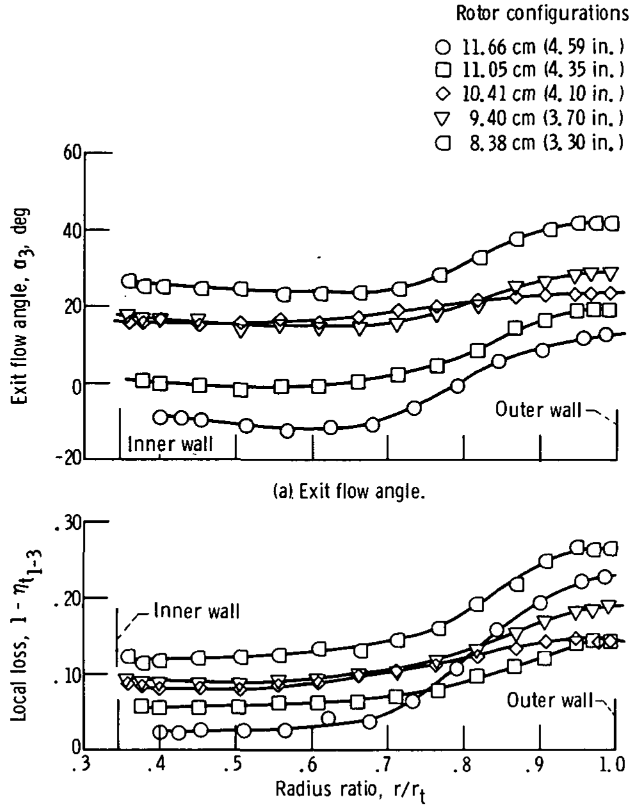


Figure 6. - Variation of rotor exit flow angle and turbine loss with radius ratio at equivalent design speed and pressure ratio.



(a) Exit flow angle.

(b) Local loss.

Figure 7. - Variation of rotor exit flow angle and turbine loss with radius ratio at equivalent design speed and at the pressure ratio corresponding to maximum efficiency.

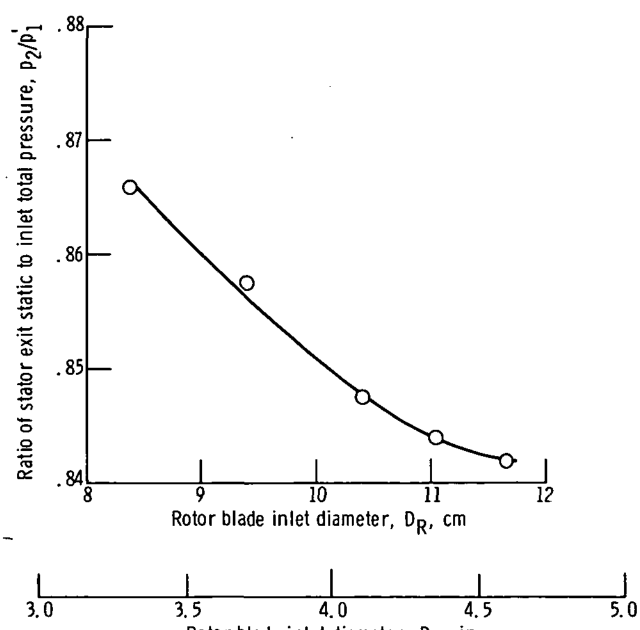
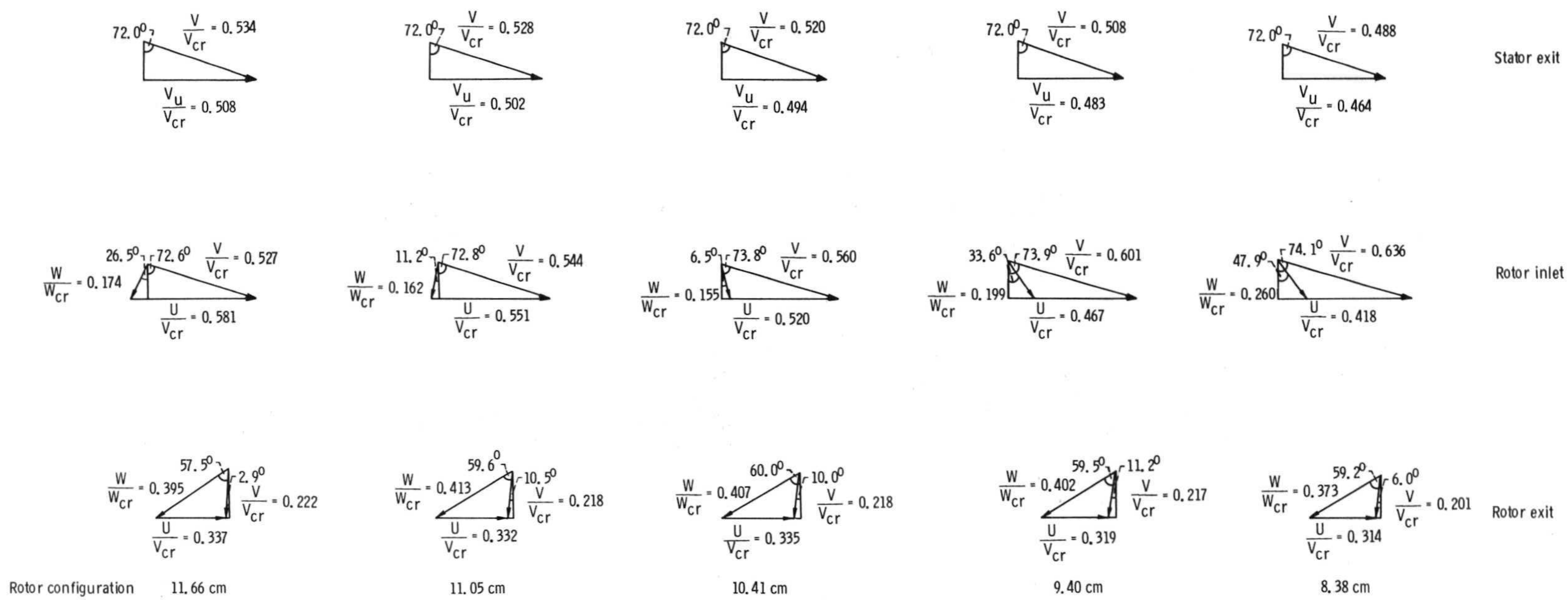
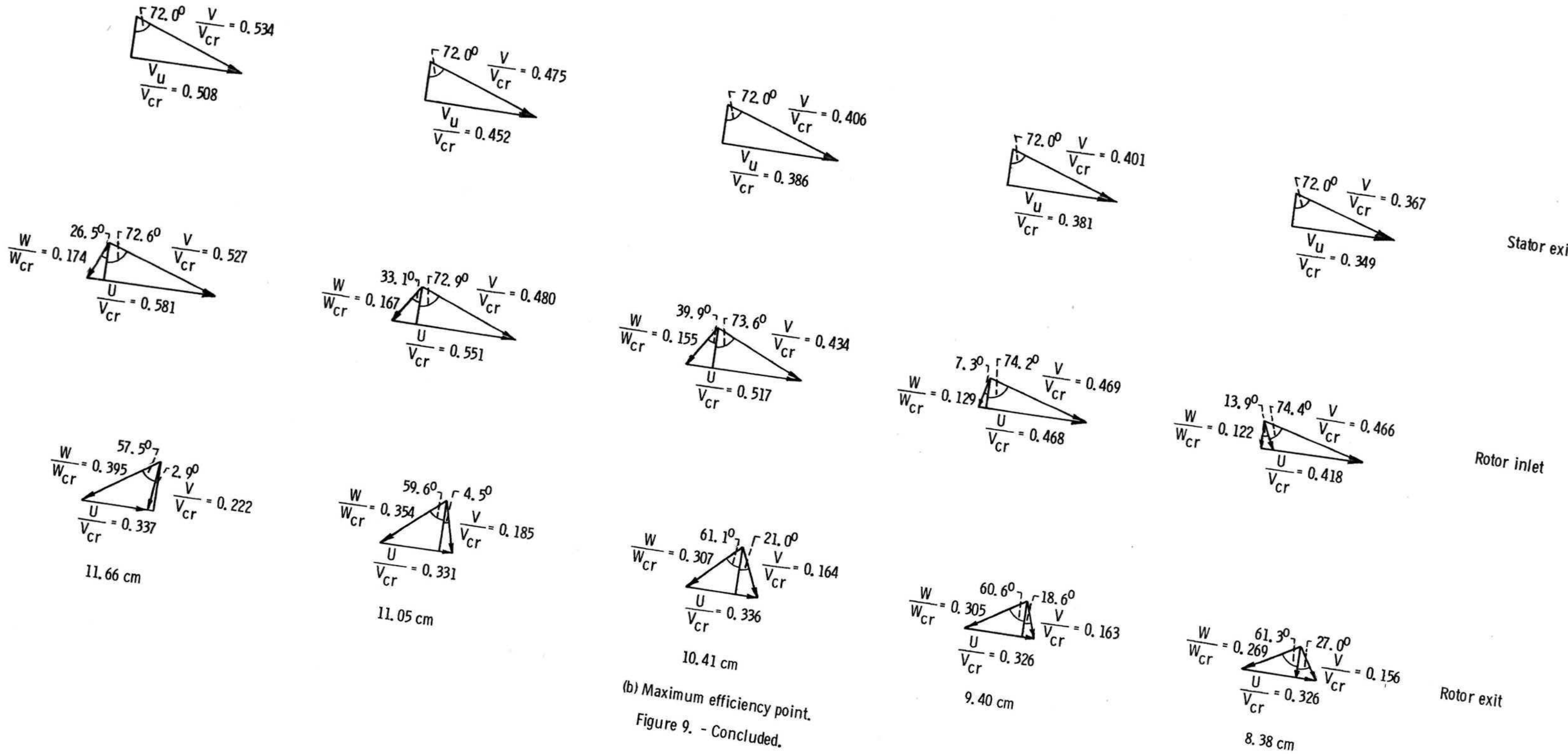


Figure 8. - Variation of stator exit static pressure with rotor blade inlet diameter at equivalent design rotative speed and pressure ratio.



(a) Design equivalent speed and pressure ratio.

Figure 9. - Turbine free-stream velocity diagrams at 50 percent streamline.



(b) Maximum efficiency point.

Figure 9. - Concluded.

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